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A LIQUID COOLER MODULE WITH CARBON FOAM FOR ELECTRONICS COOLING APPLICATIONS (PREPRINT)

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Electrochemistry and Thermal Sciences Branch
Power Division

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14. ABSTRACT
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A Liquid Cooler Module with Carbon Foam for Electronics Cooling Applications

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Abstract

A liquid cooler module (LCM) employing a high thermal-conductivity, pitch-based carbon foam is studied. The newly developed carbon foam has an effective conductivity up to 150 W/m-K and porosity up to 90%. It is believed that this high-conductivity carbon foam could significantly enhance the heat transfer due to the thermal dispersion effect. To prove the concept of the carbon foam liquid cooler, a three-dimensional numerical study of the carbon foam cooler was undertaken. The numerical results indicated that even with a heat flux as high as 100 W/cm^2, the average temperature drop between the substrate and the liquid coolant is less than 20 ^{o}C. Experimental study was also undertaken for the LCM and the results were compared with the numerical results.

Nomenclature

c specific heat, J/kg-K
C_2 coefficient in the momentum source term
F inertia coefficient constant
K permeability of porous medium, m^2
k conductivity, W/m-K
$\dot{m}$ mass flow rate, kg/s
p pressure, N/m^2
q heat flux, W/cm^2
S_i momentum source term
T temperature, ^{o}C
V volume, m^3
$\vec{V}$ velocity vector
u, v, w, x-, y-, and z-direction velocities, respectively, m/s
x, y, z, x-, y-, and z-coordinates

Greek symbols

ε porosity
μ fluid dynamic viscosity, kg/m-s
ρ density, kg/m^3

Subscripts

b base of substrate
e effective
f fluid
i fluid inlet
l liquid coolant
m mean average
p porous medium
t total

Introduction

A commonly used method of cooling power electronics onboard aircraft is liquid convection cooling by employing polyalphaolefin (PAO) as the coolant (Lin et al., 2002). The cooling device includes a dielectric silicon or aluminum compound substrate, a copper fin array soldered onto the substrate, and a lid with coolant tube manifolds. The fin array consists of a number of plain fin strips and has discrete contact areas on the substrate. The fin strips can be offset or aligned in the flow direction with a small gap between two consecutive fin strips. Excellent heat transfer results were obtained with the aforementioned fin array cooler (Lin et al., 2002). For mass flow rates greater than 0.077 kg/s, the cooler with the offset fin strip layout can handle a heat flux level of 46 W/cm^2 with a substrate-to-coolant temperature difference smaller than 37.5 ^{o}C.

The objective of this paper is to further study the fin array cooler described above by incorporating the newly developed high thermal conductivity, pitch-based carbon foam (Klett, 1998) into the cooler. In this study, the copper fin array is replaced with the carbon foam, and the heat transfer characteristics of this liquid cooler are investigated. The results are compared to those of the fin array cooler. The goal is to minimize the substrate-to-coolant temperature difference at the same or a higher heat flux levels reported by Lin et al (2002).

The high thermal conductivity, pitch-based carbon foam was developed by Oak Ridge National Lab (Klett,

1998) based on a novel production method. The mesophase pitch-derived graphitic foam is dimensionally stable with a low coefficient of thermal expansion (CTE), which is generally close to that of an aluminum compound substrate. With a porosity of 75-90%, the effective conductivity of the foam is still as high as 150 W/m-K. The foam is also very lightweight, with a specific gravity (density) of 0.58 (g/cm^3) as compared to 8.9 (g/cm^3) for copper.

Heat transfer augmentation using a foam-material filled duct was studied by Hunt and Tien (1988a, 1988b). Their work indicated that a heat transfer increase by several times is achievable with the deployment of the foam-material system. They believed that the heat transfer augmentation is due to the so-called thermal dispersion effect. The dispersion process is caused by the presence of the solid matrix which forces the flow to undergo local transverse motion. This local transverse motion of the fluid results in an additional radial flux that augments the heat transfer. At higher flow rates, the dispersion dominates the molecular diffusion and therefore, the thermal conductivity of the fluid has little effect on the heat transfer. This is a useful result especially to dielectric fluids which have poor thermal conductivity. Hwang and Chao (1994) studied the heat transfer in sintered porous channels. The channels were made of sintered bronze beads with two different mean diameters, d_p = 0.72 and 1.59 mm, which result in effective thermal conductivities of 10.6 W/m-K and 10.3 W/m-K, respectively, for the two porous samples. Using air as the coolant, they found that the forced air heat transfer coefficient could be increased from 0.01 to 0.5 W/cm-K. Lee and Howell (1991) studied the heat and mass transfer coefficients around a porous medium put on a flat plate at a distance from the leading edge of the flat plate both numerically and experimentally. The permeability of ceramic blocks used for experiment was experimentally determined in conjunction with Forchheimer equation. More recent studies related to fluid flow and heat transfer in porous media may be found in Faghri and Zhang (2006).

With the utilization of a foam material having a very high thermal conductivity, it is anticipated that the high heat flux from the wall could be effectively spread to the entire solid matrix of the foam. Since a foam system could provide a solid-liquid contact area many times greater than the wall surface, the total heat transfer rate could be significantly increased. In addition, the wall temperature and the temperature difference between the wall and the coolant could be substantially reduced.

Numerical Analysis

The proposed liquid cooler employing the high-conductivity carbon foam is first studied numerically. The configuration of the cooler is shown in Fig. 1, which is similar to that used by Lin et al. (2002) except that the copper fin array is replaced with a carbon foam of the same size. The conservation equations for mass, momentum, and energy in a porous medium subject to the assumption of local thermal equilibrium between the porous medium and the fluid flow are:

$$\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}+\frac{\partial w}{\partial z}=0 \qquad (1)$$

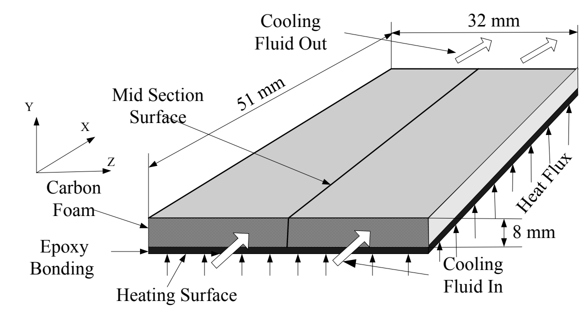

Fig. 1 Schematic of carbon foam cooler

$$\frac{\rho}{\varepsilon^2}(\vec{V}\cdot\nabla)\vec{V}=-\nabla p-\left[\frac{\mu}{K}+\frac{\rho F}{\sqrt{K}}\left|\vec{V}\right|\right]\vec{V}+\frac{\mu_e}{\varepsilon}\nabla^2\vec{V} \qquad (2)$$

$$(\rho c)_f(\vec{V}\cdot\nabla T)=k_e\nabla^2 T \qquad (3)$$

where

$\left|\vec{V}\right|=(u^2+v^2+w^2)^{1/2}$, and

$k_e=\varepsilon k_f+(1-\varepsilon)k_p$

The above expression for the effective conductivity is a simple but widely used one in many areas including the heat pipe area (Faghri, 1995). However, it should be recognized that the expression by nature is an approximation for the thermal property of a porous medium filled with a liquid. Although there are some more sophisticated alternative relations available in the literature (Faghri and Zhang, 2006), further validation of the relations with available experimental data may be needed. In addition to the assumption of the local thermal equilibrium between the porous medium and the fluid, the above governing equations imply that the porous medium is assumed to be homogeneous and isotropic and the velocity is the local volume-averaged Darcian velocity. The liquid flow is also assumed steady and incompressible. The porosity is defined as the ratio of pore volume, V_p, to the total volume of the porous medium, V_t:

$\varepsilon = V_p / V_t$

To accurately evaluate the effective viscosity, μ_e, a model may be needed. However, it has been found that taking $\mu_e \approx \mu$ provides a good agreement with experimental data (Lee and Howell, 1991). Therefore, this assumption is adopted in this study.

The flow and heat transfer in the carbon foam in conjunction with the heat conduction in the substrate are solved using a CFD software, FLUENT. In the FLUENT, porous media are modeled by addition of a momentum source term to the standard fluid flow equations. The source term is composed of two parts, a viscous loss term and an inertial loss term and is of the following form

$$S_i = \left(\frac{\mu}{\alpha} v_i + C_2 \frac{1}{2} \rho |v_i| v_i\right) \tag{4}$$

Comparing the above equation with Eq. (2), it is clear that

$$\alpha = K, C_2 = \frac{2F}{\sqrt{K}} \tag{4a}$$

For a porous medium, the permeability K and inertia coefficient constant F generally need to be determined through experimental data. Consider a simple, one-dimensional problem, the Forchheimer equation can be written as

$$-\frac{dp}{dx} = \frac{\mu}{K} u + \frac{\rho F}{\sqrt{K}} u^2$$

The above equation can be rewritten as follows (Lee and Howell, 1991):

$$-\frac{1}{u}\frac{dp}{dx} = A + Bu \tag{5}$$

where

$$A = \frac{\mu}{K}, \quad B = \frac{\rho F}{\sqrt{K}}$$

Due to extremely complex geometries of the solid matrix in the carbon foam, it is believed that sufficiently accurate analytical expressions for both permeability, *K,* and inertia coefficient constant, *F,* may not be readily available. Therefore, it was decided that these two parameters would be determined indirectly using the data from the experimental study that would follows. For a given number of experimental data related to mass flow rate and pressure drop, the best values for *A* and *B* can be found through the method of least squares. Once the values of *A* and *B* are found, *K* and *F* can be evaluated from the above two relations. Experiments have been conducted to find the relation between the fluid velocity and the pressure drop across the carbon foam, which will be described later. Using a set of experimental data in conjunction with the method of least squares, the values of *K* and *F* were found to be $K = 1.28 \times 10^{-10}\ m^2$ and $F = 1.2 \times 10^{-4}$, respectively. Once the values of *K* and *F* are determined, the value of C_2 can be evaluated using Eq. (4a).

Results of the Numerical Analysis

With the determined values of *K* and C_2, extensive numerical calculations were undertaken with the coolant mass flow rate varying from 0.01 kg/m^2 to 0.05 kg/m^2, the heat flux at the bottom surface of the carbon foam varying from 26 W/cm^2 to 100 W/cm^2, and a coolant inlet temperature of 74.5^{o}C. The porosity and effective conductivity of the carbon foam were taken to be 0.8 and 100 W/m-K, respectively. Figures 2 - 8 illustrate the velocity and temperature distributions on an *x-y* plane of the carbon foam with *x* representing the longitudinal direction of the flow and *y* representing the transverse direction. Unlike the flow in a duct, the velocity distribution in the *y* direction is substantially uniform with essentially no traditional boundary layer developing over the carbon foam top and bottom surfaces, which is a typical characteristic of flow in a porous medium (Fig. 2). The slightly higher velocity near the substrate surface is due to a higher temperature there, which somewhat reduces the fluid density and induces a higher volumetric velocity. Unlike the velocity distribution, thermal boundary layer develops over the heating surface. As the mass flow rate is increased, the thermal boundary layer over the heating surface becomes thinner. In general, the thermal boundary layer is very thin within the mass flow rate range simulated (Figs. 3-5 and Figs. 6-8), which results in a very effective heat transfer. In any cases, the maximum temperature at the heating surface is below 400 K. For the purpose of comparison, a case without the carbon-foam was numerically simulated, and the result is presented in Fig. 9. As can be seen, the maximum temperature at the heating surface has reached nearly 1,400 K. The result indicates that the carbon foam is very effective for the reduction of the heating surface temperature.

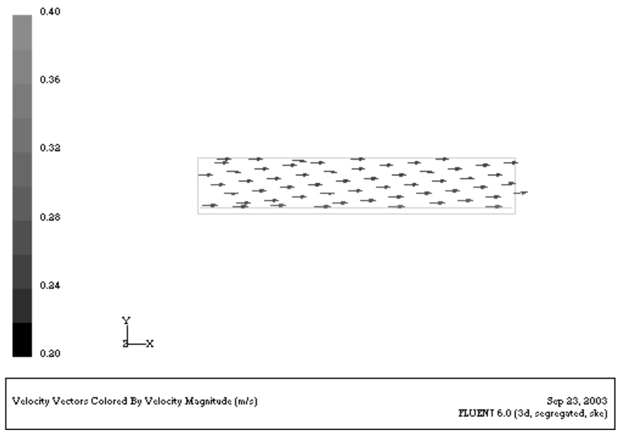

Fig. 2 The velocity distribution at mid section surface for q = 100 W/cm^2, $\dot{m}$ = 0.05kg/s . Velocity range: 0.24 ~ 0.36 m/s

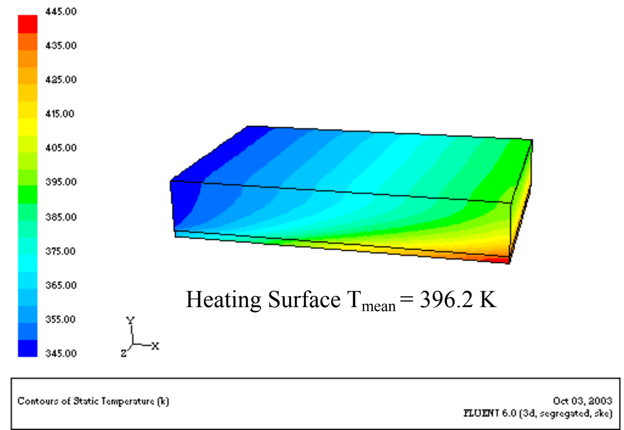

Fig. 5 The temperature distribution for q = 100 W/cm^2, $\dot{m}$ = 0.01kg/s .

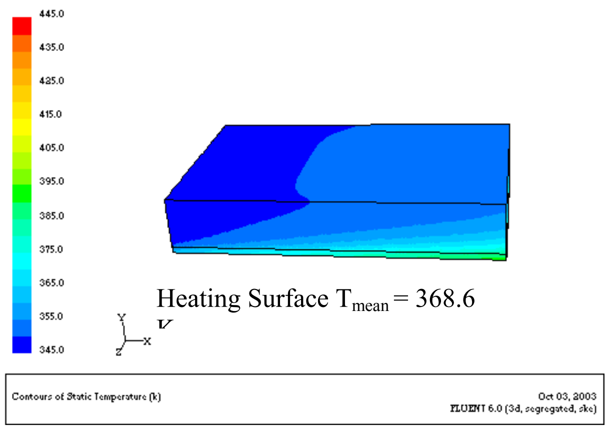

Fig. 3 The temperature distribution for q = 100 W/cm^2, $\dot{m}$ = 0.05kg/s .

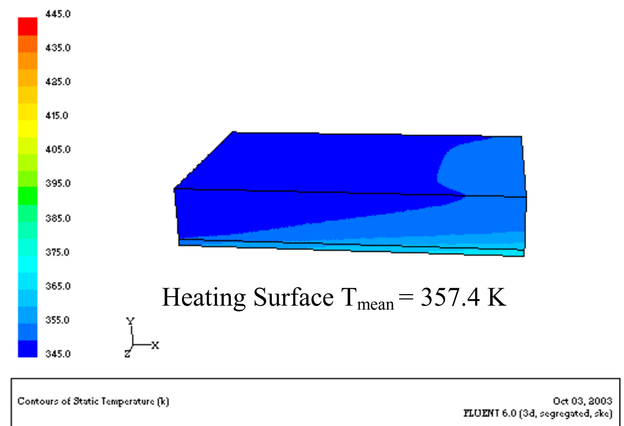

Fig. 6 The temperature distribution for q = 46 W/cm^2, $\dot{m}$ = 0.05kg/s .

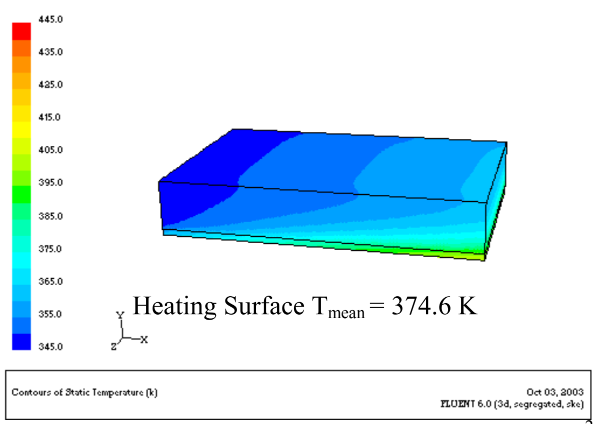

Fig. 4 The temperature distribution for q = 100 W/cm^2, $\dot{m}$ = 0.03kg/s .

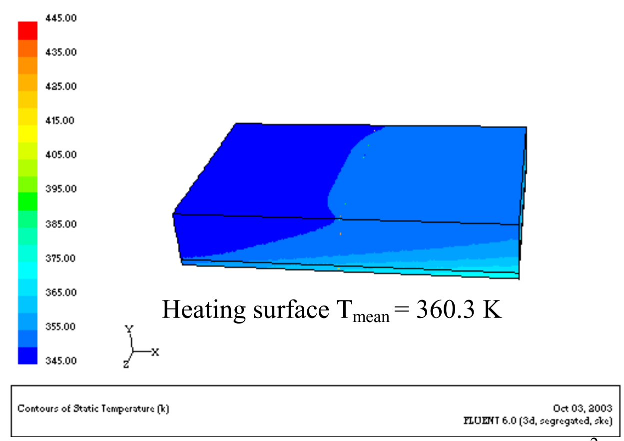

Fig. 7 The temperature distribution for q = 46 W/cm^2, $\dot{m}$ = 0.03kg/s .

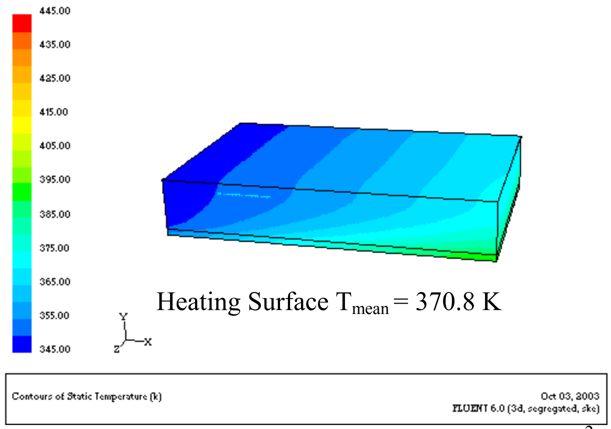

Fig. 8 The temperature distribution for q = 46 W/cm^2, $\dot{m} = 0.01$kg/s.

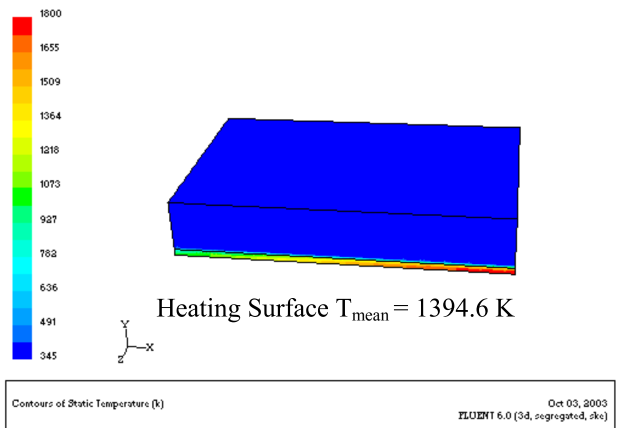

Fig. 9 The temperature distribution for q = 100 W/cm^2, $\dot{m} = 0.05$kg/s without the foam.

One of the important parameters which can be used to gauge the performance of the cooler is the temperature difference between the substrate base (heating surface) and the coolant, ΔT. Figure 10 plots this temperature difference $T_{b,m} - T_{l,m}$ vs. the mass flow rate $\dot{m}$, where $T_{b,m}$ is the average temperature at the substrate base in contact with the carbon foam and $T_{l,m}$ is the average coolant temperature. The same data are also tabulated in Tables 1-3. Along with the numerical results, some experimental data for the off-set fin array cooler are also plotted for the purpose of comparison. As can be seen from Fig. 10 and Tables 1 - 3, the carbon foam significantly enhanced the heat transfer in the cooler and could handle a heat flux as

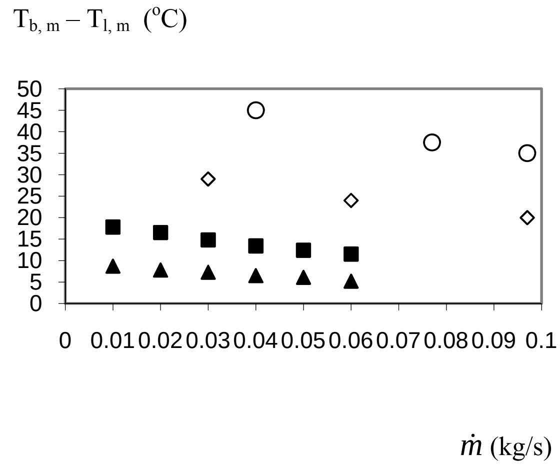

Figure 10. Substrate-to-coolant temperature difference versus mass flow rate. ▲ q = 46 W/cm^2, carbon foam cooler; ■ q = 100 W/cm^2, carbon foam cooler; ◊ q = 26 W/cm^2, fin array cooler; experimental; ○ q = 46 W/cm^2, fin array cooler, experimental

Table 1. Numerical results for carbon foam cooler ($T_{l,i}$ = 347.5K, coolant: PAO, q = 46 W/cm^2)

Flow Rate (Kg/s)	$T_{b,mean}$ (K)	$T_{l,mean}$ (K)	ΔT (K)	ΔP (bar)	$T_{b,max}$ (K)	$T_{b,mean}$(K)
0.01	371.9	363.3	8.6	0.38	387.1	376.8
0.02	363.4	355.4	8.0	0.88	372.0	368.2
0.03	359.7	352.6	7.3	1.1	366.8	364.6
0.04	358.2	351.5	6.7	1.8	364.2	363.1
0.05	356.9	350.7	6.2	2.2	362.3	361.8
0.06	355.9	350.2	5.7	2.7	360.9	360.8

Table 2. Numerical results for carbon foam cooler ($T_{l,i}$ = 347.5K, coolant: PAO, q = 100 W/cm^2)

Flow Rate (Kg/s)	$T_{b,mean}$ (K)	$T_{l,mean}$ (K)	ΔT (K)	ΔP (bar)	$T_{b,max}$ (K)	$T_{b,mean}$ (K)
0.01	398.3	380.4	17.9	0.24	428.0	408.8
0.02	380.4	363.9	16.5	0.6	400.1	390.9
0.03	373.0	358.2	14.8	1.0	388.3	383.5
0.04	368.9	355.4	13.5	1.4	381.6	379.4
0.05	366.2	353.8	12.4	1.7	377.4	387.5
0.06	364.2	352.7	11.5	2.1	374.4	384.5

Table 3. Numerical results for carbon foam cooler ($T_{l,i}$ = 323K, coolant: PAO, q = 26 W/cm^2)

Flow Rate (Kg/s)	$T_{b, mean}$ (K)	$T_{l, mean}$ (K)	ΔT (K)	ΔP (bar)	$T_{b, max}$ (K)	$T_{b, mean}$ (K)
0.015	334.0	329.3	4.7	0.8	340.6	336.8
0.02	332.2	327.7	4.5	1.1	337.5	334.9
0.03	330.2	326.1	4.1	1.7	334.3	332.9
0.04	329.1	325.3	3.8	2.3	332.6	331.8
0.05	328.3	324.9	3.4	3.0	331.4	331.1
0.06	327.8	324.5	3.3	3.5	330.6	330.5

high as 100 W/cm^2 with a very moderate temperature drop between the substrate and the coolant. An important factor that could limit the application of the carbon foam cooler is the pressure drop, Δp, between the inlet and outlet of the carbon foam cooler. When the mass flow rate is relatively high, the pressure drop Δp also could be substantially high (Tables 1 - 3). As a result, the carbon foam cooler should be designed to work at a moderate mass flow rate. It should be pointed out that the numerical results presented here represent the maximum heat transfer enhancement achievable under ideal working conditions. The substrate temperature was considered the same as that on the bottom surface of the carbon foam. In real situation, these two temperatures could be substantially different due to a finite contact thermal resistance between these two surfaces.

Experimental Study

For the experimental study, a carbon foam test sample was prepared. The test sample consists of an aluminum nitride ceramic substrate and a block of carbon foam bonded to the substrate, as shown in Fig. 1. The carbon foam is bonded to the substrate using a silver-filled epoxy, Aremco-Bond 556. The carbon foam is the high thermal conductivity, pitch-based carbon foam from Oak Ridge National Lab. The overall size of the carbon foam block that is bonded to the substrate is 51.0 mm (length)×33.0 mm (width)×8.0 mm (height). The experimental setup for the present study is based on that used by Lin et al. (2002), which is schematically shown in Fig. 11, for measuring heat transfer rates, temperature distributions on the substrate surface, and pressure drop across the carbon foam. The fin array assembly in Fig. 11 was replaced with the carbon foam-substrate assembly that was installed in a housing and the coolant flow path was created by closing a lid onto the top surface of the housing. For

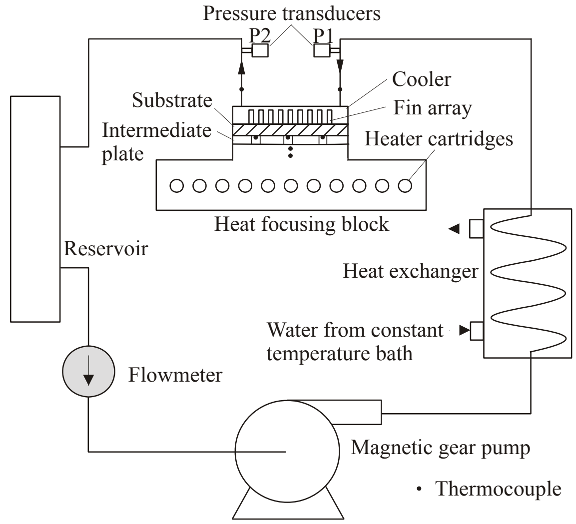

Figure 11 Schematic of the experiment setup (Lin et al., 2002)

more detailed description of the experimental setup, the reader is referred to the paper of Lin et al. (2002). During the testing, the heat transfer rate, *Q*, into the carbon foam and the temperature distribution at the interface between the carbon foam and the substrate are measured. As shown in Fig. 11, eight thermocouples were embedded in 0.7 mm holes drilled along two planes in the upper part of the heat focusing block. The heat transfer rate can be calculated by using the average temperature difference between the two planes where the thermocouples were located, the distance between the two planes, and the cross-sectional area of the focusing block between the two planes. To measure the temperature at the bottom surface of the substrate, 10 thermocouples were soldered onto the surface of the substrate. The locations of these 10 thermocouples are shown in Fig. 12. To protect the thermocouples from contacting the heater, a 0.8 mm thick intermediate plate was used between the substrate and the heat block (Lin et al., 2002). The intermediate plate was soldered onto the substrate which has been replaced by a copper plate. In the present study, the substrate is aluminum nitride ceramic. After the intermediate plate was soldered to the substrate, the assembly of the substrate and the intermediate plate significantly deformed due to a thermal mismatch between the ceramic substrate and the copper intermediate plate. To overcome this difficulty, the intermediate plate was not soldered to the substrate. Instead, the intermediate plate was pressed between the heat focusing block and the substrate with thermal grease applied between the substrate and the intermediate plate and between the intermediate plate and the heat block. With this method, substrates with a variety of materials having different thermal expansion

coefficients can be tested on the experimental setup. Once the temperature at the bottom surface of the substrate was measured, the temperature on the substrate upper surface (on the flow side) can be calculated using the one-dimensional heat conduction law in conjunction with the known heat transfer rate Q.

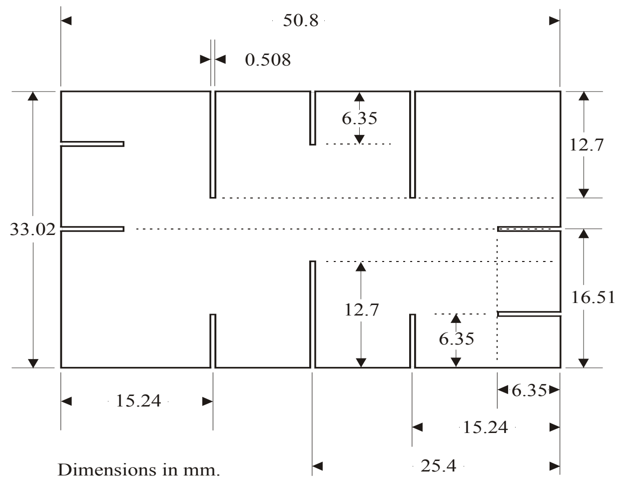

Figure 12 The thermocouple locations on the substrate and intermediate plate (thickness: 0.8 mm) used to protect the thermocouples

Experimental Results and Discussion

Experiments were conducted for the flow rates from 0.015 kg/s to 0.025 kg/s, heat inputs from 13 W/cm^2 to 26 W/cm^2, and the coolant inlet temperatures from 50 to 74.5 ^oC, and the results are tabulated in Table 4. It can be seen from Table 4 that the temperature drops between the substrate and the coolant, which is represented by $T_{b,m} - T_{l,m}$, are substantially higher than the corresponding numerical result. The cause of this high temperature difference was investigated, and it was identified that the low conductivity of the epoxy caused a large temperature drop between the substrate and the carbon foam. To facilitate a firm bonding between the substrate and the carbon foam, a relatively thick layer of epoxy was applied between the carbon foam and the substrate due to the small fraction of solid matrix on the surface of the foam in contact with the substrate. If a 1 mm thick of epoxy is assumed and the highest thermal conductivity value provided by the vendor is used, the temperature drop across this layer of epoxy would be $(25\,W/cm^2)(0.001\,m)/(9.39\,W/m-K) = 26.6^oC$, for a given heat flux of 25 W/cm^2. Since the thermal conductivity of the epoxy could be substantially lower than that provided by the vendor, an even higher temperature drop could be possible. As a result, a much higher temperature difference between the substrate and the coolant was generated during the experiment. To eliminate this interfacial temperature drop, the carbon foam test sample should be prepared using a better bonding technique or the carbon foam should be processed over the solid substrate surface instead of being epoxy bonded to a solid surface.

Table 4. Experimental results for the carbon foam cooler sample

Flow Rate (Kg/s)	$T_{b,\,mean}$ (K)	$T_{l,\,mean}$ (K)	ΔT (K)	ΔP (bar)	$T_{l,\,i}$ (K)	q (W/cm^2)
0.015	375.9	348.2	27.6	0.87	345.6	13.2
0.20	374.8	374.9	26.87	1.21	346.0	13.3
0.025	373.3	347.7	25.6	1.7	346.1	13.3
0.015	379.6	326.7	52.93	1.02	321.5	26.0
0.02	377.3	325.7	51.55	1.49	321.7	26.4
0.025	375.5	325.0	51.8	2.07	321.8	26.7
0.015	381.7	335.9	45.81	0.93	331.5	22.2
0.02	379.5	335.0	44.5	1.30	331.7	22.2
0.0245	377.4	334.7	42.71	1.63	331.9	22.65
0.015	390.6	345.1	45.43	0.84	341.0	21.7
0.02	389.2	345.0	44.3	1.01	341.7	21.9
0.0245	387.7	344.6	43.13	1.37	342.0	22.1

Conclusions and Suggested Future Studies

A literature survey indicated that the heat transfer in a single phase flow could be significantly improved through the utilization of a foam material due to the thermal dispersion effect. With the newly developed high thermal-conductivity, pitch-based carbon foams, it is anticipated that a liquid cooler employing the carbon foam could significantly enhance the heat transfer and achieve a higher heat flux condition. A three-dimensional numerical study has been undertaken to prove the concept of a liquid cooler based on the high-conductivity carbon foam. The numerical results indicated that the carbon foam based liquid cooler worked very well. At a heat flux of 46 W/cm^2, the average temperature drop between the substrate and the coolant is below 10^{o}C. Even at a heat flux of 100 W/cm^2, the average temperature difference is still below 20^{o}C. A liquid cooler with such an enhanced heat transfer characteristic could be a potential candidate for cooling power-electronic devices onboard aircraft. An experimental study was also conducted for the carbon foam liquid cooler. A test sample was prepared by bonding a block of carbon foam to an aluminum nitride ceramic substrate with a silver-filled epoxy. The sample was tested on a previously developed experimental setup with a modified intermediate-substrate assembly. The modified

assembly achieved accurate temperature reading on the bottom surface of the substrate without the need to solder the intermediate plate onto the substrate. As a result, substrates with a variety of materials having different thermal expansion coefficients could be tested on the experimental setup. Experimental data at different working conditions were collected. The experimental results indicated that the temperature differences between the substrate and the liquid coolant were much higher than those from corresponding numerical results. It was believed that these high temperature differences were caused by the existence of the low-conductivity epoxy that was used to bond the carbon foam to the substrate. Further experimental studies are needed using new test samples. The new sample should be prepared by a better bonding technique or by processing the carbon foam directly on a solid metal surface.

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